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SAVINGS BY DESIGN
FIELD VERIFICATION AT A RURAL PUBLIC SCHOOL

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Field monitoring was performed at a rural public primary school (Commercial PMD No.C200027) to characterize the winter peak demand savings of the building operating with a ground-coupled heat pump system, over a base-case resistance-heated building. An incentive was provided for these savings under Ontario Hydro's Savings by Design commercial incentive program. The predicted savings, using design input data and the LoadShaper computer model, were 108 kW. Based on the measured data and on the reanalysis of the simulation, the actual peak demand savings are estimated to be 22 kW. This large overprediction was mainly due to the building being built and operated differently from its preliminary design. Problems with one of the LoadShaper algorithms and low performance of the heat pump system accounted for the balance of the overestimation. However, there were still significant measured savings because of the heat pump system installed in the school, showing the importance of this technology for Demand Management purposes.

INTRODUCTION

In October 1990, Ontario Hydro awarded a $32,400[1] Savings by Design program incentive to the school board for installation of a ground coupled heat pump (GCHP) system in the expanded building. The Savings by Design program promotes energy efficient designs in commercial buildings by offering financial incentives for buildings in the design stage or those undergoing a major retrofit.

The Energy Analysis Group in the Technical Services and Development Department performed a simulation using LoadShaper /1/, a technical software package. They determined a one-hour peak demand savings of 108 kW for a GCHP improved case over a resistance heating base case for this building. The peak demand for the base case was 207 kW, while that of the improved case was 99 kW.

[1] @ the 1990 rate of $300/kW saved

Field monitoring was conducted during the 1991 winter/spring season. This study was performed to determine the actual peak demand savings available in operation, for comparison with the predicted savings.

BUILDING DESCRIPTION

The school is a one storey building with 8 classrooms (Kindergarten to Grade 6), a lunch room, a gymnasium, a library and administrative offices, covering about 1800 m^2. It is largely new construction added to the original two-classroom baseboard-heated building.

During the winter months, heating is provided to the building from fourteen ground coupled heat pumps connected to a glycol loop running throughout the school. The loop gets its energy from ground coils buried under the playing field. A rooftop ventilation heat recovery unit (HRU) draws exhaust air from the washrooms and passes it through a heat exchange coil to warm the incoming outdoor air before it enters the ventilation ductwork into the school. Note that the heat recovery unit was common to the base case and the improved design.

MONITORING

Spot measurements were performed in February, March and May to determine the general performance of the heat pump and heat recovery systems. Data obtained from longer term monitoring (February to May) were used to extrapolate the performance levels to colder outdoor temperatures. The parameters calculated from these measurements were the steady state Heat Pump Performance Levels (HPPLs, described in the next paragraph), the steady state coefficients of performance (COPs), and the ventilation heat recovery efficiencies.

Heat Pump Performance Levels (HPPLs) were calculated as the ratio of thermal output on the air side, to the local electrical input to power the blower and compressor of the specific heat pump. The output was based on the measured air flow rates and temperature differences across the unit. Note that the HPPL differs from the COP; the latter is the ratio of total thermal output to total electricity consumed (ie, including measured glycol circulation pumping power).

HRU efficiency is the ratio of heat actually recovered from the exhaust stream to that theoretically available to be recovered /2/. It was based on the measured outdoor air, exhaust air, and warmed ventilation air temperatures, and ventilation air flow rates.

Equations for HPPL, COP and HRU efficiency can be found in Appendix A.

RESULTS AND DISCUSSION

a) i) Steady State HPPL

Based on two sets of spot measurements on one of the heat pumps (HP 13) in February, its steady state HPPL was 2.3$\pm$0.3, compared to the design specifications of 3.2. To confirm this lower than expected value, a similar performance level of 2.2$\pm$0.2 was obtained for both the same heat pump and another one (HP 8), from measurements taken in May (with the heating mode locked in for 15 minutes before measurements were taken).

The longer term monitored data of HP 13 were analyzed. Data from two days with different ambient temperatures were chosen. The daily average, steady state HPPLs for February 21 (warmer day) and February 27 were 1.9 and 2.0, respectively. These values were slightly lower than the spot-measured values possibly due to residual part load inefficiencies. Start up losses of the heat pump compressor for the first few minutes of each "on" cycle may have caused these inefficiencies.

ii) Comparisons with ARI Ratings

The steady state COPs based on the spot-measured HPPL, the ARI ratings and the design specifications were 2.6, 3.0, and 3.2, respectively. These COPs have been adjusted, based on the manufacturer's data for different operating conditions, to match the rating conditions from the ARI Standard /3/. Differences of actual operating conditions (air and glycol temperatures, and glycol flow rate) from the rating assumptions cannot explain the low COP measured.

b) HRU Efficiency

Daily average efficiencies, extracted from the longer term monitoring, on February 20 and 25 were 55%. This is lower than the 66% based on the mechanical drawing specifications, but it matched the efficiency assumed in the LoadShaper analysis.

c) Need for Commissioning and Training of Building Operator

During the initial visit to the school on January 16, the HRU exhaust fan was not operating (and there was no indication of how long this condition had existed). At our instigation, this was fixed on February 14. On March 19, it was noted that the entire ventilation system had not been operational since March 1. Not only were the demand savings due to ventilation heat recovery not met during these periods, but this latter episode resulted in an unhealthy environment for the staff and students while the main source of fresh air was not being supplied into the school.

Based on these two experiences, it is suggested that the building operator could have been better acquainted with potential failure modes, indications of incorrect operation, and corrective procedures. The poor heat pump performance that was measured, further suggests the need for more complete commissioning of all the building's mechanical systems.

d) LoadShaper Analysis vs. Actual Operation

Because the LoadShaper analysis was based on information gathered during the design process, some of the inputs differed from the ultimate "as built" situation. The details which were not well modelled for both input information reasons and because of algorithm limitations in LoadShaper, are discussed below with indications of the effects on the expected savings.

i) GCHP Heating Demand during Pick-Up

LoadShaper requires an auxiliary heating input as it does not model a fully-sized GCHP, but only those connected to a "water" loop with a central source of heating and cooling. The LoadShaper input attempted to model this by entering an auxiliary boiler of zero capacity. Seemingly, the interaction between the zero capacity of the boiler and night setback caused a

decrease in demand during morning pick-up, an unphysical result. The resulting one-hour heating input component of the building demand was (incorrectly) calculated to be 13 kW, compared to (a correct) 129 kW for the resistance base case. Table 1 shows the original LoadShaper winter peak heating profile obtained for the base case and the improved case.

When LoadShaper was run with night setback and a full-capacity auxiliary boiler for the improved case, an input of 44 kW was calculated (with no input needed by the auxiliary boiler). This makes physical sense as it shows an increase in demand during pick-up. The last column of Table 1 shows this peak heating profile. This revised improved case had a total building demand of 126 kW. Thus 27 kW of putative saving were due to the algorithmic error when setback and zero auxiliary boiler capacity were combined.

ii) Night Setback

As indicated above, night setback was assumed for both base and improved LoadShaper runs. This creates two problems. Firstly, night setback amplifies the difference between the base and improved cases for the one-hour peak period considered. Savings by Design uses a one-hour peak period to match the period used for demand billing. But the avoided costs to our generation system, upon which our system benefit is based, are incurred over the 16-hour period from 07 00 to 23 00 on winter working weekdays. Thus, using the one-hour peak amplification due to pick-up after setback is inappropriate for assessing demand savings. The Energy Analysis Group has not included a setback component in their analyses since mid-1990 for just such reasons.

The second difficulty is that night setback should not be used with a GCHP system and in fact, is not used at the school.

The LoadShaper files were altered again to exclude setback and yielded peak demands of 152 kW and 109 kW for the base and improved cases, respectively, for a net savings of 43 kW. LoadShaper overpredicted an additional 38 kW of savings by assuming night setback.

iii) GCHP COP

In the LoadShaper analysis, the glycol pump was entered as "auxiliary equipment to the heat pumps". In this way, the COP of 3.45 that was entered was really an HPPL, with the added glycol pumping power compensating for the difference between the HPPL and COP. A further improved case was run by substituting a realistic heating COP of 2.5[2] based on the steady state spot measurements. The peak demand for the improved case is then 116 kW. This suggests that the demand savings were overpredicted by 7 kW due to this COP problem.

During the peak hour, the space heating inputs for the base and improved cases were 75 and 38 kW, respectively. These results suggest a peak hour operating COP of 2.0 (=75 kW/38 kW). This is lower than the steady state rating because a) the glycol temperature at peak is colder

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[2] LoadShaper input requires the ratio at 8.3°C (47 F) vs. the ARI standard 10°C (50 F) entering glycol temperature used in section a)i)

(1.7°C vs. 8.3°C) and b) the peak hour load is less than the heat pump capacity so that a part load efficiency is included.

From data collected on February 21 and 27, the heating demands between 08 00 and 09 00 were 17 and 19 kW, respectively. These values, when compared to the 38 kW from the further improved case, seem reasonable since these days were about 10°C warmer than the peak day assumed in the LoadShaper analysis.

iv) Ventilation Heat Recovery and Preheat

Because a heat recovery module is unavailable in LoadShaper, the amount of ventilation air per person was adjusted from 11 to 5 cfm/person in the LoadShaper files. This energy-adjusted flow rate compensated for the heat that would be recovered by a 55% efficient HRU. However, both the ventilation and exhaust fan powers (nominally 5.5 kW each) are simultaneously reduced by the same 5/11 factor in the modelling but not in reality. This causes a reduced load to be calculated for the building and results in a slight overprediction in the calculated savings by about 1.5 kW (recalling that LoadShaper suggests an operational COP of 2.0 at peak conditions).

Operation of the 40 kW ventilation air preheater in the HRU is also not able to be accounted for in the current version of LoadShaper. It is likely, under the preheater's control scheme, that the full 40 kW will be on during the building's demand peak. Now, after heat recovery, the design ventilation load from LoadShaper is just 43 kW. Thus most of that load is actually met resistively, whereas the analysis has it met with the GCHP. This inflates the predicted savings by about 20 kW.

v) Lighting

During the period from 07 00 to 18 00, the LoadShaper input lighting level was 53 kW, while the effective actual lighting level for this same period was 42 kW from the electrical drawings. Figures 1a) and 1b) indicate the LoadShaper lighting profile and the actual profile (from the custodian), respectively. Although lighting power is lower, the contribution to space heating assumed in LoadShaper must be compensated for. Thus the difference of 11 kW is supplied by the heating systems. The resultant savings were underpredicted by 5.5 kW.

vi) Measured Ventilation Flow Rates

The original design called for 2.9 m^3/s based on 11 cfm/person for 560 persons. During the Savings by Design process, the number of persons expected was changed to 415 and so the flow rate was scaled down to 2.1 m^3/s. However, the fan size was not changed in reality and was still designed to deliver the 2.9 m^3/s.

The measured ventilation air flow rate was 3.2 m^3/s. (There are in fact only 315 occupants, yielding a healthful rate of 21 cfm/person.) This 50% higher ventilation load (over the original 43 kW load) results in an 11 kW underestimation of the demand savings.

vii) Number of Building Occupants

As mentioned previously, there are 100 fewer students in the building than predicted (315 instead of 415), providing about 5 kW less metabolic heat. This results in a 2.5 kW underprediction of the demand savings.

viii) Baseboard Heaters

Space heating by the heat pumps is supplemented by baseboard heaters in one classroom of the original school, at the entrances to the building and in the washrooms. Possibly, high transmission losses from the large window area made it necessary to operate the three baseboard heaters in the classroom at full capacity during most of the winter season.

Assuming the peak heater load was 62% of the installed heater load (25 kW out of a possible 40 kW), the original LoadShaper analysis, which assumed no baseboard heater operation, overpredicted the savings by 12.5 kW.

ix) Glycol Circulation Pump

The LoadShaper analysis used 5 kW for the glycol pump. The measured power was 3.14 kW. During the morning peak hour, 1.86 kW less power is needed to operate the pump, but 1.86 kW more heat is needed for space heating. This results in about 1 kW less power needed overall (ie, 1 kW overprediction) when that heat is supplied with a COP of 2.0.

As suggested above in section d) iii), the heat pump system would be modelled better by entering real COP values in the simulation, and no extra equipment such as circulation pumps.

Table 2 summarizes each of the above effects on the total building peak demand savings. The result is a net overprediction in savings of 86 kW. Hence, the audit, monitoring, and reanalysis of the operation of the school indicate peak demand savings of 22 kW, rather than the 108 kW originally estimated. Few of the reasons for this discrepancy were within the control of the Energy Analysis Group at the time when the Savings by Design analysis was performed.

The difference between predicted vs. actual savings for this building may be "worse" than other projects, but the issues it raises are probably generally applicable. Demand savings based only on design information may not match the savings in actual operation. Unless there is some confirmation that the building operates as it was designed, Ontario Hydro may not get the system peak savings that it is paying for. It seems that it would be worthwhile to pay for commissioning etc. to ensure real savings.

e) Comparison of Monitored/Reanalyzed Operation to Billing Data

The peak billing demand of the school was 128 kW in January. This is different than the previously calculated demand of 160 kW for the improved case. There are three areas in which the design calculation does not match the billing peak conditions.

- LoadShaper assumed weather for a Toronto winter design day. The date of the billing peak and thus the actual temperature at the peak is unknown. If it was warmer than assumed in LoadShaper, then part of the billing vs. LoadShaper difference is explained. But it is likelier

that peak conditions were colder (the design temperature is 5°C colder than in Toronto; this is not too important for the improved versus base case comparison).

- The exhaust fan of the HRU was probably not operating at the peak. There was no power requirement from this fan, nor was there any significant heat recovery. Therefore, the heat pumps completed tempering of the ventilation air, almost compensating with their COP of 2.0 for the 55% efficient HRU. The 5.5 kW fan power input reduction changes the LoadShaper result towards the billing value.

- The baseboards in the existing classroom were activated for comfort reasons after the billing peak. Thus the heat requirement was met at peak with the heat pump, lowering the electrical input required.

The above factors tend to explain the 160 kW versus 128 kW difference. Note that the former value is our best estimate of the long-term operating characteristic of the school. Our inability to fully explain the difference (for lack of information) somewhat reduces the confidence with which we estimate the 22 kW savings from the GCHP.

CONCLUSIONS

1. Based on the audit, monitoring and reanalysis of the system operation of the school, the winter peak demand savings totalled about 22 kW. Although these savings are lower than the 108 kW from the original estimate, the savings due to installation of the heat pumps are still quite substantial.

2. The main reasons for the above discrepancy are:

   a) inappropriate LoadShaper algorithm for modelling of the ground coupled heat pump (GCHP),

   b) a steady state GCHP COP of 2.6, lower than the ARI rating of 3.0 and the manufacturer's specification of 3.2,

   c) other operational characteristics different from design values, and

   d) the inability of LoadShaper to model the ventilation heat recovery unit and its preheater.

3. Operation of energy efficient systems may not reflect their design specifications if they are not fully commissioned and if the operators are not fully trained.

RECOMMENDATIONS

1. The Savings by Design Program should be expanded to allow funding of operational checks on systems receiving incentives. Payment of all of the incentive could be contingent on actual performance meeting design specifications, within reasonable accuracy limits.

2. The Board of Education that governs this school should be informed of the operational deficiencies found in this project and urged to make sure that the mechanical systems of the

building operate within reasonable limits of design specifications. The building operator should be more fully acquainted with potential failure modes, indication of incorrect operation and corrective procedures.

3. Morgan Systems (creator of LoadShaper) should be made aware of the limitations to properly modelling this building's mechanical systems as found in this project.

4. The Energy Analysis Group should be allowed to audit designs presented for Savings by Design verification so as to suggest improvements. In this specific case, it could have recommended that the HRU preheater and heat pump temperature control schemes be integrated to allow heating of the ventilation air primarily by the heat pump system except at very low ambient temperatures.

5. The Energy Analysis Group should better document analysis difficulties, as this information may be pertinent to analyzing future Savings by Design projects.

6. The Energy Analysis Group should be given some time free of deadline pressure to more closely check their own analyses, especially when analyzing novel energy efficient systems.

7. To make better use of limited time and resources, a future monitoring project should be approached in the following order:

   - determine the specific objective(s) of the project
   - obtain LoadShaper files and do enough supplementary analysis to understand the major operational features of the building
   - perform a walk-through audit of the building and speak with the building operator
   - obtain billing data to compare with the LoadShaper file analysis and building operator's information
   - if required, perform spot and longer term monitoring
   - present/report findings.

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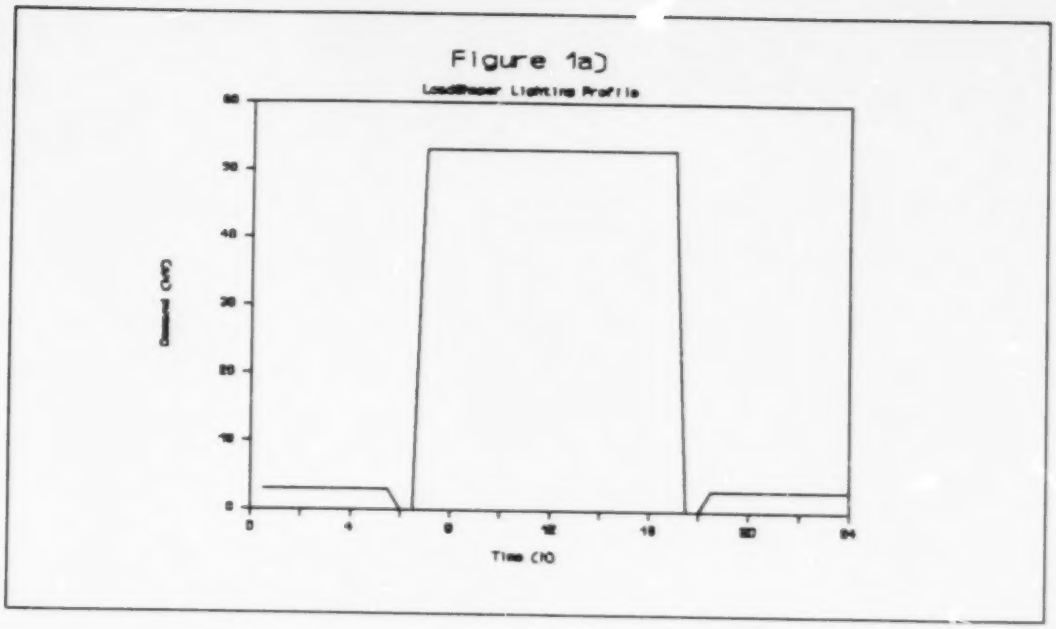
Figure 1a)
LoadShaper Lighting Profile
Demand (kW)
Time (Hr)

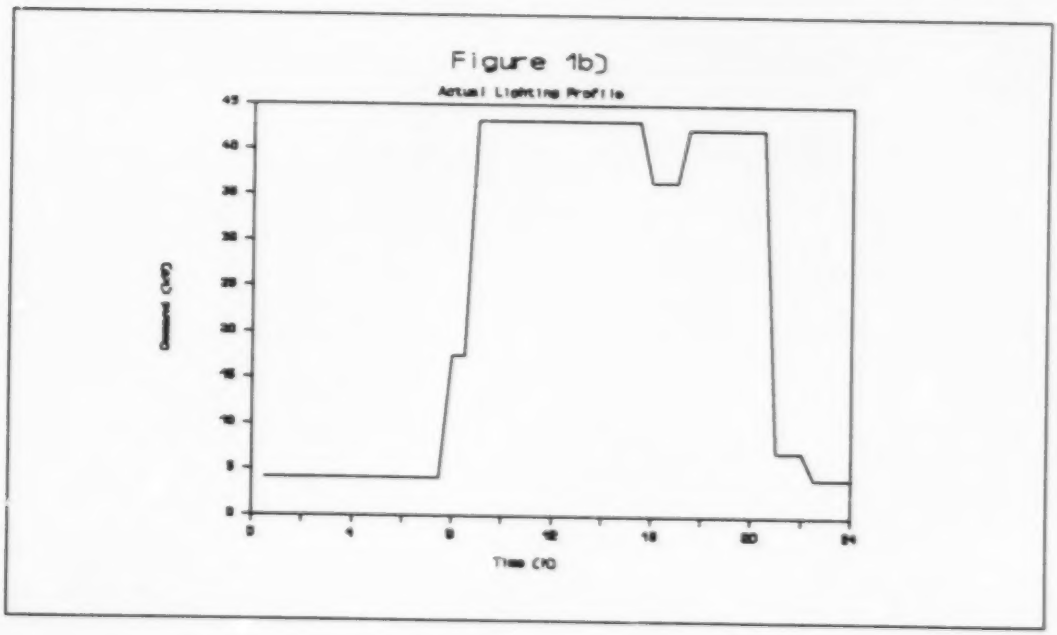
Figure 1b)
Actual Lighting Profile
Demand (kW)
Time (Hr)

Table 1 - LoadShaper Peak Heating Inputs

Hour	Original Base Case Heating Input (kW)	Original Improved Case Heating Input (kW)	Adjusted Improved Case Heating Input (kW)
01 00	48	27	18
02 00	51	27	19
03 00	54	27	20
04 00	56	28	21
05 00	58	28	21
06 00	60	28	22
07 00	61	28	22
08 00	54	24	20
09 00	129	13	44
10 00	111	13	40
11 00	99	12	36
12 00	89	11	32
13 00	87	10	31
14 00	75	9	27
15 00	71	9	25
16 00	69	9	25
17 00	76	9	27
18 00	0	17	0
19 00	5	18	2
20 00	24	24	9
21 00	30	25	11
22 00	35	26	13
23 00	41	26	15
24 00	44	27	16

Table 2 - Effects of Changes to LoadShaper files on Peak Hour Demands and Respective Demand Savings

Change to LoadShaper file		Peak Hour Demand Base Case (kW)	Peak Hour Demand Improved Case (kW)	Demand Savings (kW)
Original cases		207.0	99.0	108.0
i)	improved case: fully-sized aux. boiler	207.0	126.0	81.0
ii)	no night setback	152.0	109.0	43.0
iii)	improved case: heating COP 2.5	152.0	116.0	37.0
iv)	add HRU preheat	152.0	136.0	16.0
	less HRU fan power	155.0	140.5	14.5
v)	less lighting	155.0	135.0	20.0
vi)	more ventilation	177.0	146.0	31.0
vii)	fewer occupants	182.0	148.5	33.5
viii)	improved case: baseboard heaters	182.0	161.0	21.0
ix)	improved case: less pumping power	182.0	160.0	22.0

Appendix A

Steady State HPPL

$HPPL = (1.2*1*V*dT)/(HPpower)$

where:	HPpower=	electricity consumed by HP compressor & fan (kW)
	1.2=	density of air (kg/m^3)
	1=	specific heat capacity of air ($kJ/kg°C$)
	V=	air volumetric flow rate (m^3/s)
	dT=	supply to mixed return air temperature difference (°C)

$V = vA$; where: v = average duct velocity (m/s)
A = duct area (m^2)

Steady State COP

$COP = [HPPL*(I - G) + G]/I$

where: $HPPL = (O - G)/(I - G)$

$COP = O/I$

and O = total thermal output from heat pump system (kW)
G = glycol pump contribution (kW)
I = total electricity input (kW)

HRU Efficiency

$HRUeff = Vs/Vmin*[(X1-X2)/(X1-X3)]$

where:	HRUeff=	heat recovery unit efficiency (%)
	Vs=	total ventilation supply air flow rate (m^3/s)
	Vmin=	lesser of the total supply and total exhaust air flow rates (m^3/s)
	X1=	enthalpy of outdoor air flowing into HRU (kW)
	X2=	enthalpy of supply air flowing out of HRU and into the school (kW)
	X3=	enthalpy of the exhaust air flowing out of the school and into the HRU (kW)

X1, X2, and X3 include the sensible and latent components.